

Development of the Heat Exchanger for the 64-m Antenna Hydrostatic Bearing

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Maintenance of oil temperature, as a means of viscosity control, is an essential requirement for the 64-m antenna hydrostatic bearing. Operational experience with the heat exchanger used for cooling the oil showed that it was not functioning adequately or in accordance with the design, and a resultant study showed a probable internal structural failure. A new heat exchanger was designed, with JPL assistance on the structural problem, and is now operating satisfactorily.

I. Introduction

The entire rotating weight of the 64-m antenna [approximately 2.5×10^6 kg (5.5×10^6 lb)] is carried on a pressurized hydrostatic bearing. In this bearing a fixed flow of pressurized oil is forced through the sills between three pads, which support the rotating weight, and the bearing runner, mounted on the concrete pedestal. The pressure required to force the fluid through the sill also acts upward to support the load on the pads. The height of the oil film between the pads and the runner is a function of the oil viscosity, and hence of the oil temperature. In operation this film height is critical, and a close control of the oil temperature is required. The oil is supplied by a system of precharge and high-pressure pumps (Fig. 1), and since no external work is accomplished by the pressurized oil, the entire motor output must be dissipated as heat. The heat loss rate, based on actual operating pressures, is approximately 105 kW (361,200 Btu/h) with one precharge pump operating and 117 kW (401,400 Btu/h)

with both precharge pumps operating. Loss through the reservoir walls is negligible. Loss through the runner into the pedestal concrete varies with the season, with a minimum of 39 kW (132,600 Btu/h) total, assuming a diurnal average temperature of 27°C (80°F) and an oil temperature of 31°C (88°F). Hence, the heat exchanger must dissipate from 66 to 78 kW (228,600 to 268,800 Btu/h).

II. Heat Exchanger Parameters

The basic relationship giving the capacity of a heat exchanger is

$$q = F \cdot A \cdot U \cdot T_{lm} \quad (1)$$

where q is the heat dissipation in watts; F is a factor depending on the heat exchanger configuration and the operating temperatures; A is the heat exchanger area; and U is an overall heat transfer coefficient dependent on the heat exchanger design, the flow rates of the oil and cool-

ant, and on the viscosity of the oil and coolant. T_{lm} is the log mean temperature difference given by the equation

$$T_{lm} = \frac{\theta_a - \theta_b}{\ln \theta_a / \theta_b}$$

where θ_a is the temperature difference between the inlet oil and its surrounding coolant, and θ_b between the outlet oil and its surrounding coolant.

For the hydrostatic bearing system the oil flow and specific heat are constant. Equation (1) can then be rewritten

$$\Delta t = \frac{F \cdot A \cdot U \cdot T_{lm}}{Q \cdot S} \quad (2)$$

Δt is the oil temperature change, Q is the oil flow, and S is the specific heat of the oil.

The quantity $Q \cdot S / (F \cdot A \cdot U)$ is then an operational parameter by which the heat exchanger performance can be evaluated, based on measurements of oil and coolant temperatures. The water flow rate must be assumed constant, or measured, and the oil temperature, and hence its viscosity, must be within a reasonable range for meaningful results.

III. Original Heat Exchanger

The original heat exchanger at DSS 14 was designed for oil with a viscosity of $116 \text{ m}^2/\text{s}$ (525 SSU) and to dissipate 75 kW (267,000 Btu/h) with oil inlet and outlet temperatures of 41.5 and 38.5°C (107 and 101°F) and coolant temperatures 34.9°C (95°F) in and 36.5°C (97.8°F) out. This gives an operating parameter as outlined above of 0.50. Trouble was experienced maintaining the oil temperature during the summer, and investigation found that: (1) a modulating valve for the water flow, controlled by oil temperature, always operated in the full open position; (2) the pressure drop across the heat exchanger on the oil side had dropped from a calculated 34.4 N/cm^2 to 17.2 N/cm^2 (50 psi to 25 psi); and (3) the operating parameter was about 0.90 rather than the 0.50 designed, implying loss of heat transfer ability. Since the unit could not be readily disassembled for inspection, the manufacturer's drawings were obtained and a stress analysis made on the structural elements.

The heat exchanger is a 2-4 type, as shown in Fig. 2, with 2 shell passes for the oil and 4 tube passes for cool-

ant. The oil inlets and outlets are on opposite sides of the same end of the heat exchanger so that at that end the full pressure drop across the oil side of the unit was carried by the baffle. The baffle was 3.2-mm (0.125-in.) carbon steel with a yield strength of approximately 344 N/mm^2 (50,000 psi) while the bending stresses due to the pressure difference, carried over the 46-cm (18-in.) span, was 5350 N/mm^2 (776,000 psi). It was clear that some sort of failure had occurred permitting the oil to pass around or through the baffle, reducing both the pressure drop between oil inlet and outlet, and the effective area of heat transfer.

IV. Replacement Heat Exchanger

A review was made of the actual antenna operating conditions and a new heat exchanger was purchased to meet the following requirements:

- (1) Cooling $11.4 \text{ dm}^3/\text{s}$ (180 gpm) of oil having a viscosity of $176 \times 10^{-6} \text{ m}^2/\text{s}$ (800 SSU) from 42 to 37.7°C (108 to 100°F), using a coolant flowing up to $18.9 \text{ dm}^3/\text{s}$ (300 gpm) and 34.9°C (95°F) at the inlet.
- (2) Cooling $22.7 \text{ dm}^3/\text{s}$ (360 gpm) of oil having a viscosity of $176 \times 10^{-6} \text{ m}^2/\text{s}$ (800 SSU) from 41 to 39°C (106 to 102°F) with the same coolant flow.
- (3) Pass $22.7 \text{ dm}^3/\text{s}$ (360 gpm) of oil having a viscosity of $198 \times 10^{-6} \text{ m}^2/\text{s}$ (900 SSU) with a pressure loss not to exceed 55.2 N/cm^2 (80 psi).
- (4) Have identical piping connections to those on the original heat exchanger.

These specifications reflected the actual operating conditions, using a higher viscosity oil than originally specified, providing for operation with two precharge pumps (critical mission configuration) without excessive pressure drop, and permitting a quick change over from the original unit.

The heat transfer design of the new heat exchanger was carried out by the manufacturer and checked by JPL. It was a straightforward design using finned tubes and a slightly longer shell to meet the heat transfer and flow requirements. The manufacturer encountered a problem in the structural design, however, because of the pressure drop across the baffle which had been the problem in the initial design. The solution to this problem was suggested by JPL and consisted of using a 22-mm (0.875-in.)-thick plate of a material having a yield stress of 414 N/mm^2

(60,000 psi) and an allowable working stress under the Unfired Pressure Vessel Code of 104 N/mm² (15,000 psi). In order to minimize bending moments carried into the pressure shell, the baffle was simply supported at the pressure shell walls in a channel section welded into the wall as shown on Fig. 3. The groove along the edge of the baffle plate assures that the load will be carried into the wall at the minimum distance from the shell plate neutral axis, minimizing bending moment. This arrangement

also permits removal of the baffle and tube bundle for inspection.

The new heat exchanger has been installed and is operating satisfactorily. Pressure and temperature taps were added to the oil and coolant piping, and platinum wire thermometers have been installed. These provide the measurement accuracy necessary to maintain proper monitoring of the unit.

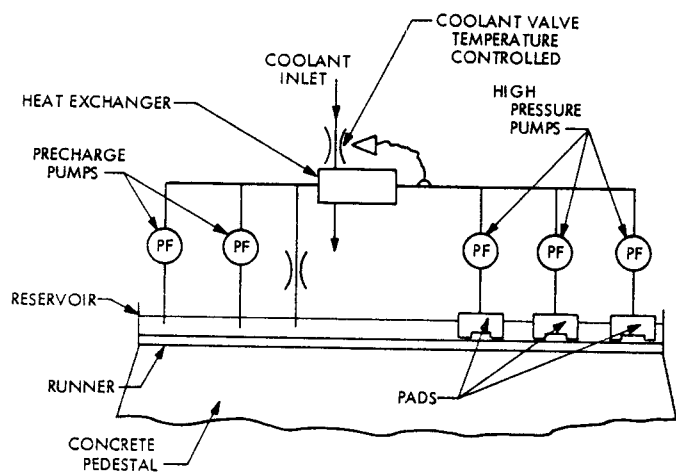


Fig. 1. Hydrostatic bearing oil cooling schematic

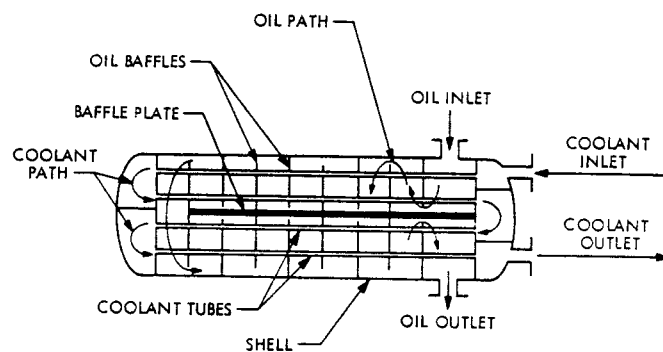


Fig. 2. Schematic of 2-4 heat exchanger

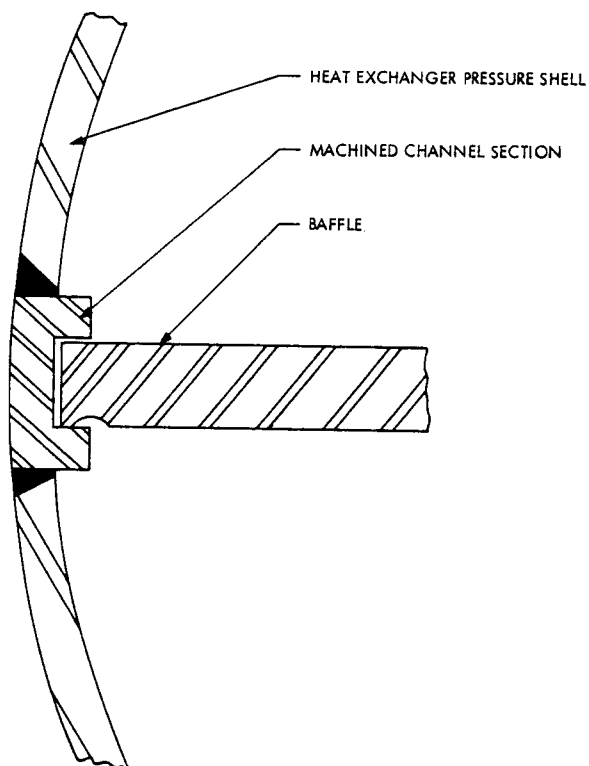


Fig. 3. Heat exchanger baffle support